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A FEASIBILITY STUDY INTO COMMERCIAL/INDUSTRIAL EXHAUST AIR HEAT PUMPS

FINAL REPORT

PREPARED FOR
ENERGY, MINES AND RESOURCES
OTTAWA, ONTARIO

JANUARY, 1987

enermodal engineering limited

energy engineers for • mechanical design • energy audits • computer analysis • product development

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EXECUTIVE SUMMARY

This report examines the thermal and economic feasibility of three heat pump heat recovery systems for commercial/industrial applications. The heat pump systems are

i) exhaust air-to-water
ii) exhaust air-to-supply air
iii) exhaust air-to-supply air with an intermediate glycol loop

The thermal and economic performance of these systems are compared to two conventional heat recovery systems: air-to-air heat exchanger and glycol run-around loop. An assessment of the potential market size is also made. The systems were compared in eleven Canadian and northern U.S. locations representing a wide range in length of heating season and electricity and natural gas prices.

In most of the locations studied the exhaust air-to-water heat pump offered the greatest energy and cost savings and the shortest payback period. Only is those locations with high electricity prices or low natural gas prices (e.g. Alberta and New England) did the glycol run-around loop offer superior performance.

The exhaust air-to-supply air heat pump had equal or slightly poorer payback times when compared to the air-to-air heat exchanger. This heat pump would appear to have its best potential when air conditioning is being added.

Despite being a very flexible system, the exhaust air-to-supply air heat pump with an intermediate glycol loop had payback periods beyond what most building owners would accept.

1. INTRODUCTION

1.1 Background

It has been estimated that approximately 600 million dollars is spent annually in Ontario to heat make-up air for industries [1]. Another way of looking at this is that this same amount of energy is being exhausted from buildings to the outdoors.

Indoor air quality is becoming a major concern in the workplace. Contaminants such as secondhand smoke, formaldehyde, radon, nitrous oxides and other inorganic chemicals are showing up in high concentrations. It is quite likely that in the future, ventilation systems will provide larger quantities of fresh air and/or will not re-circulate indoor air. These developments will result in heating of ventilation air, as opposed to process air, being a much larger cost than at a present.

At present the only method of recovering some of the energy that is exhausted from buildings is by use of a heat exchanger between incoming and outgoing air. There are several types of heat exchangers, however, the most common is a plate type air-to-air heat exchanger. Other types includes heat pipes, heat wheels and glycol run-around loops.

Air-to-air heat exchangers recover approximately a constant percentage of the temperature difference between two air streams. At high temperature differences a heat exchanger recovers large amounts of heat but at small temperature differences very little heat is recovered. As a result, most air-to-air heat exchangers are used for recovering heat from process air (temperatures significantly above room temperature) and less often for heat recovery on ventilation air (room air temperature).

The purpose of this study is to examine the technical and economic feasibility of recovering heat from exhaust air using heat pump technology (that is, an exhaust air heat pump or EAHP). This study builds on a research project undertaken for the Ontario Ministry of Energy. In that project an exhaust air heat pump was designed, built and tested. An exhaust

air heat pump would have its evaporator in the building exhaust air duct and the condenser heat would be transferred to meet either a water or air heating load. An exhaust air heat pump differs from air-to-air heatexchangers in that it recovers a relatively constant amount of energy regardless of indoor-to-outdoor temperature difference. Thus, EAHP may offer superior performance and cost-effectiveness to air-to-air heat exchangers where the heat source and sink temperatures are close together.

Although there are many possible EAHP configurations, this study will examine only three: exhaust air-to-process water, exhaust air-to-supply air and exhaust air-to-supply air with an intermediate glycol loop. An exhaust air-to-indoor air heat pump for space heating and cooling will not be examined. This system would be similar to the exhaust air-to-supply air heat pump with slightly poorer performance because of its requirement for higher condenser temperatures. (In addition, the technical and economic feasibility of a residential sized unit has already been studied [2,3].) The three EAHP systems to be examined are self-contained units with no on-site refrigeration work required. While it is possible to use a split system heat pump connected on-site with refrigeration piping, most commercial/industrial heat pumps are self-contained roof top units.

The technical and economic feasibility of the three exhaust air heat pumps will be compared with two conventional forms of exhaust air heat recovery: plate type air-to-air heat exchanger and glycol run-around loop. These conventional forms of heat recovery transfer heat from exhaust air-to-supply air only. In general, heat exchangers cannot be used to transfer heat from exhaust air to process (mains) water because their temperatures are too close together. Section 2 of this report describes in more detail each of the five heat recovery systems. Section 3 compares the thermal performance of the five systems in several Canadian and U.S. locations. Section 4 compares the installed costs and cost effectiveness of the systems. An assessment of the market size for each of the systems is given in Section 5.

1.2 Review of Exhaust Air Heat Pumps on the Market

A review of trade journals and discussions with consulting engineers revealed that there is no commercial/industrial heat pump system specifically designed and marketed for use with exhaust air. The BEST corporation of Mississauga has custom-designed and built several exhaust air-to-water heat pump systems. They do not, however, produce a packaged unit; each system is custom designed.

There are several exhaust air heat pumps available for the residential market. Fibreglass Canada Inc. is currently marketing the "Habitair", a Swedish built exhaust air-to-water heat pump. During the heating season heat is recovered from exhaust air and transferred via the heat pump to the domestic hot water tank. A fan coil connected to the DHW tank allows partial space heating. When cooling is required, the ducts are switched so that heat is extracted from the indoor air and transferred to the domestic hot water tank. The exhaust air is not conditioned in the summer mode. DEC International Inc. of Madison, Wisconsin produce a similar unit called Therma-Vent.

Cambridge Manufacturing of Cambridge, Ontario produce a space and DHW heat pump system called the "Peach". This unit is mounted inside the house with large supply and return ducts connected to the outdoors. Outdoor air is circulated through the unit. Dampers in the unit allow indoor and outdoor air to mix together before the evaporator. Thus the evaporator sees a temperature higher than just the outdoors. This unit also has the option using mains or well water as a heat source for the evaporator.

A Danish heat pump, called the Nilan, transfers heat from exhaust air to supply air much like an air-to-air heat exchanger. This product is being field-tested at the Flair homes ENERDEMO project in Winnipeg. This product may soon be on the market under license by a Canadian company. The Mechanical Engineering Division of the National Research Council have developed and tested a similar product.

Hooper and Angus (Toronto) and Keeprite Industries (Brantford, Ontario) have developed an exhaust air space heating heat pump [2,3]. When space heating is required, air is exhausted from the house across the heat pump evaporator. Air leaks into the house to replace the exhausted air. The heat pump condenser is placed in the heating system ductwork. Keeprite has yet to commercialize this product.

All of the products described above are designed and sized for single-family residential buildings. Their maximum exhaust flow rate is generally limited to under 100 l/s (200 CFM). Commercial/industrial exhaust air system flow rates are generally 500 to 10000 l/s (1000 to 20000 CFM) and sometimes larger. Clearly these residential products are not suitable for commercial/industrial systems.

2. THE HEAT RECOVERY SYSTEMS

2.1 Overview

This study examines five exhaust air heat recovery systems for commercial/industrial applications:

1) exhaust air-to-water heat pump (EA-W-HP)
2) exhaust air-to-supply air heat pump (EA-SA-HP)
3) exhaust air-to-supply air with an intermediate glycol loop (EA-G-SA-HP)
4) exhaust air-to-supply air heat exchanger (ATA-HE)
5) glycol run-around loop

In order to compare these systems, it is necessary to select the type of building that they will be installed in. For the purposes of this study, it has been assumed that the exhaust air is handled by a rooftop exhauster, supply or make-up air is handled by a rooftop unit and any heat recovery equipment would be retrofitted to these systems. In most commercial/industrial buildings where mechanical ventilation is required, air handling equipment is placed on the roof as shown in Figure 2.1. The major exception to this situation would be high rise office buildings. In these buildings fresh air is often brought in through vertical grills in intermediate floors. Because of their short hours of operation, office buildings are probably not a good application for exhaust air heat recovery.

2.2 System Descriptions

2.2.1 Exhaust Air-to-Water Heat Pump

A drawing of the exhaust air-to-water heat pump is shown in Figure 2.2 and of a typical installation in Figure 2.3. The heat pump could be mounted inside or outside the building. It is probable, however, that it would be easier and cheaper to install the unit on the roof. The heat pump's large size would make it difficult to get through doorways and its large weight would require extra support for the ductwork.

Fig. 2.1

Typical Exhaust Air Heat Recovery Application Site

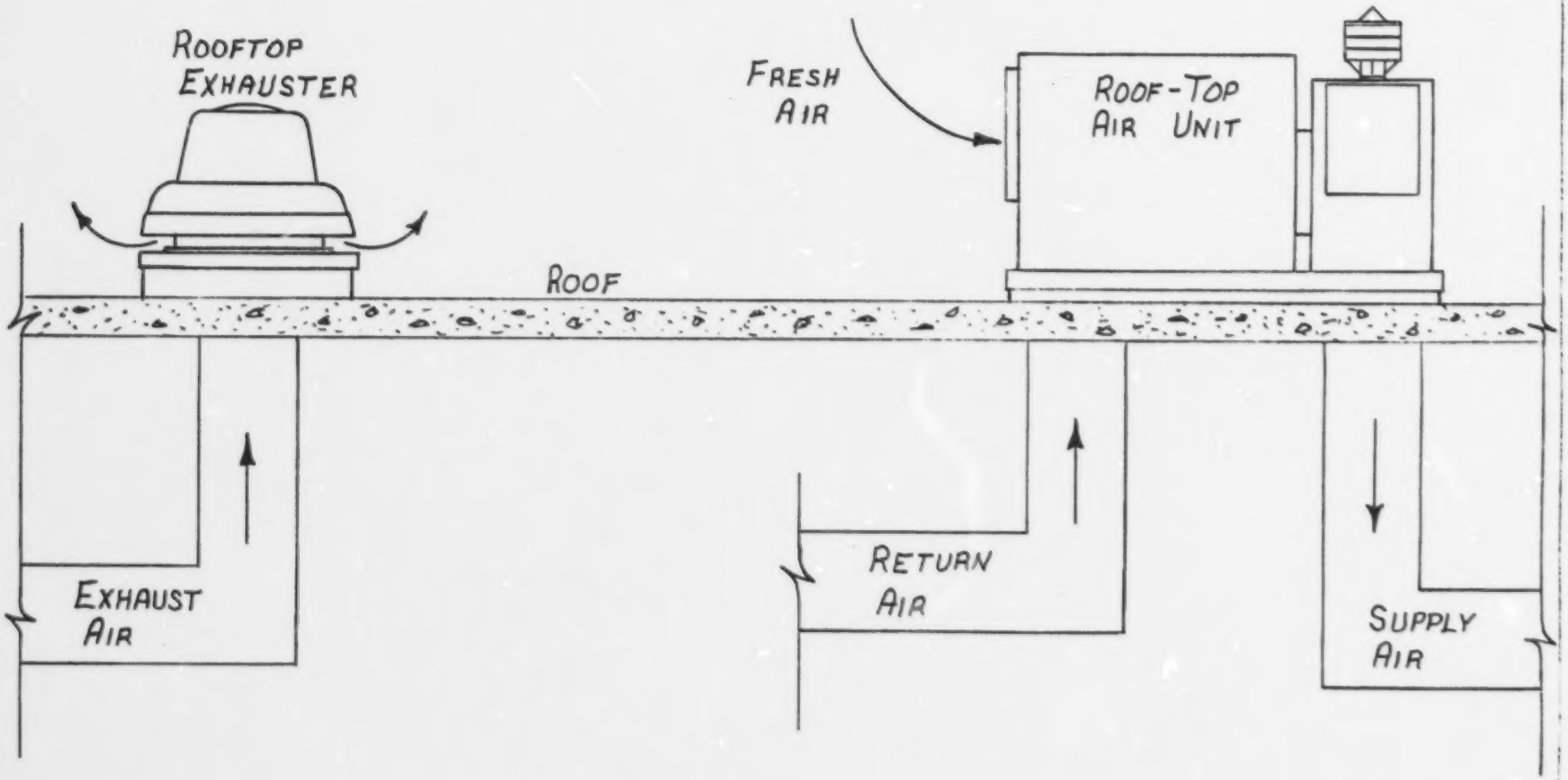
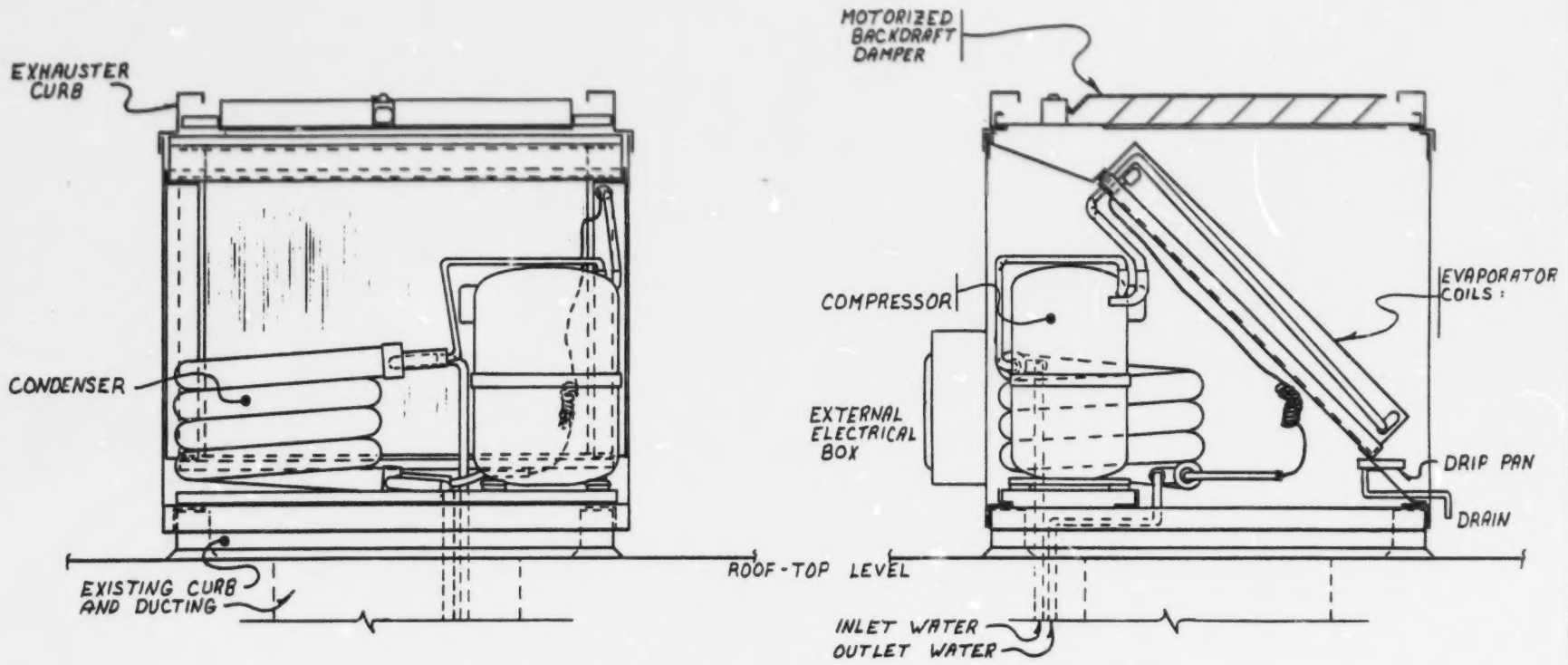


Fig. 2.2

ROOF EXHAUSTER TO BE PLACED ON TOP OF UNIT, SEALED AND SECURED TO THE 'EXHASTER CURB'.



NOTES:

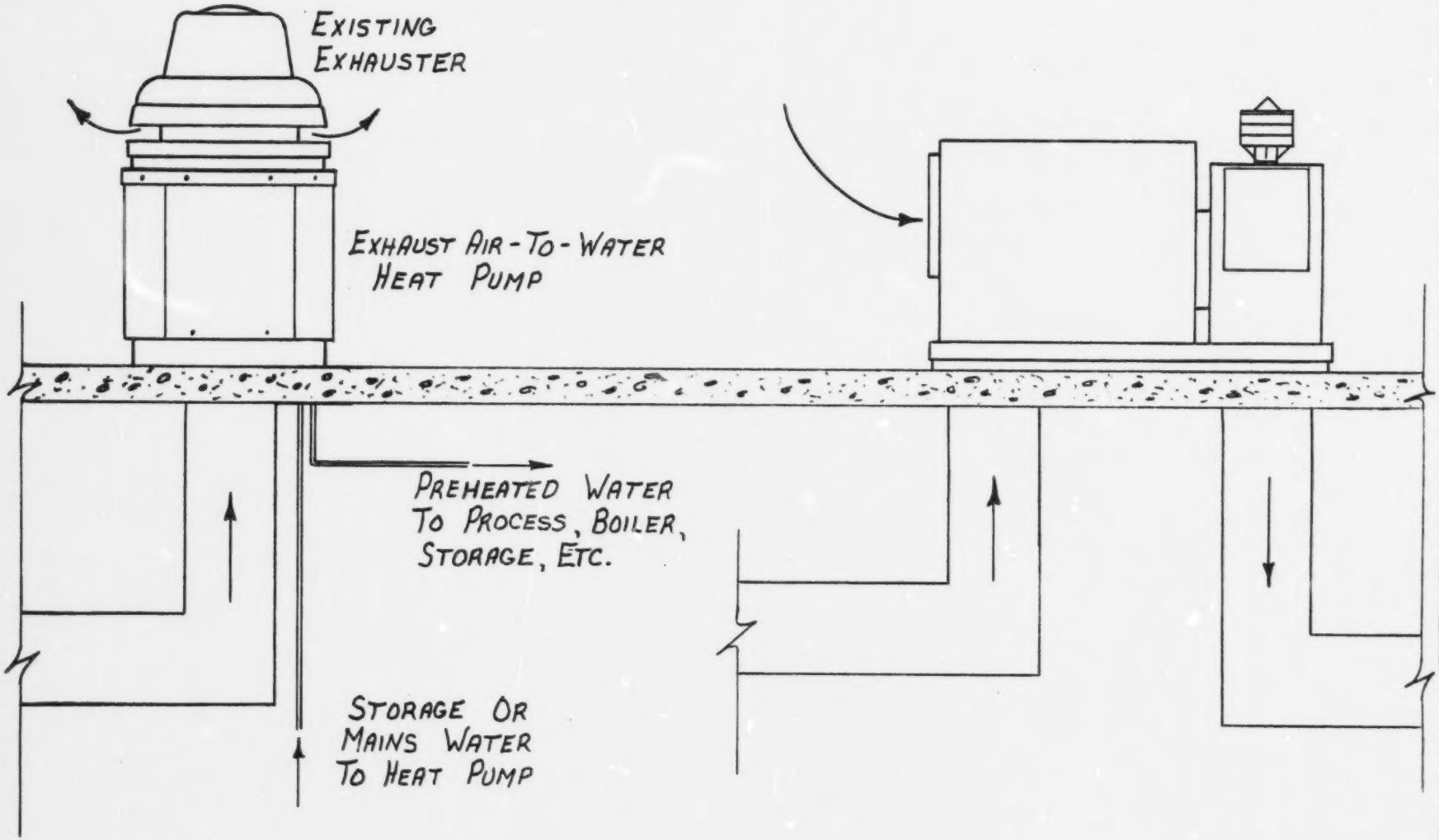
- DRAWINGS SHOWN WITH FRONT CABINET SURFACE REMOVED

- NOT SHOWN:
  - 13 mm INSULATION ON ALL CABINET SIDE PANELS
  - OPTIONAL INLET AIR FILTER

TITLE: EXHAUST AIR TO WATER ROOF-TOP HEAT PUMP		
DWN. FOR: ENERMODAL ENG'G LTD.		DWN. BY: A. HANSEN
SCALE: N.T.S.	DATE: JULY 9, 1986	DWG. NO.: EA-W-001

Fig. 2.3

Application Of An Exhaust Air-To-Water Heat Pump


Existing Exhauster
Exhaust Air-To-Water Heat Pump
Preheated Water To Process, Boiler, Storage, Etc.
Storage Or Mains Water To Heat Pump

The evaporator pressure drop can be kept low by slightly oversizing the coil and having a wide fin spacing. In this manner the existing exhauster fan should be capable of overcoming the additional coil pressure drop by changing the fan pulleys. The evaporator coil would be inclined to allow condensation to drain to a drain pan.

For applications where there is a relatively constant hot water requirement, the water lines would be connected in series and as a preheat to the conventional water heater. If the water load was not constant a small storage tank and pump would be required. The water lines are protected from freezing by heat tracing wire and the indoor air flow. During a power failure, the insulated cabinet interior temperature would be maintained at the building interior temperature by the natural buoyancy of the warm building air.

2.2.2 Exhaust Air-to-Supply Air Heat Pump

The exhaust air-to-supply air heat pump (Figure 2.4) is similar to the previous system except that the condenser is a coil. Since the objective of this study is to examine packaged systems, insulated ductwork is used to connect the heat pump to the make-up air unit instead of refrigerant piping. When the building requires cooling the compressor would be shut off. For those buildings without an air-conditioning system, the heat pump could be reversed to provide cooling. A fan would probably be required in the supply air section of the heat pump to overcome the extra friction associated with the coil and the long ductwork.

2.2.3 Exhaust Air-to-Supply Air Heat Pump with an Intermediate Glycol Loop

This system would use the exhaust air-to-water heat pump and insulated glycol piping to connect to the make-up air unit (Figure 2.5). A coil would be placed over the air intake. The coil would have to be larger than the air intake because of the flow area blocked by the coil tubes and fins. As with the previous system the compressor would be shut off during the cooling season or reversed to provide air-conditioning.

Fig. 2.4

Application Of Exhaust Air-To-Fresh Air Heat Pump

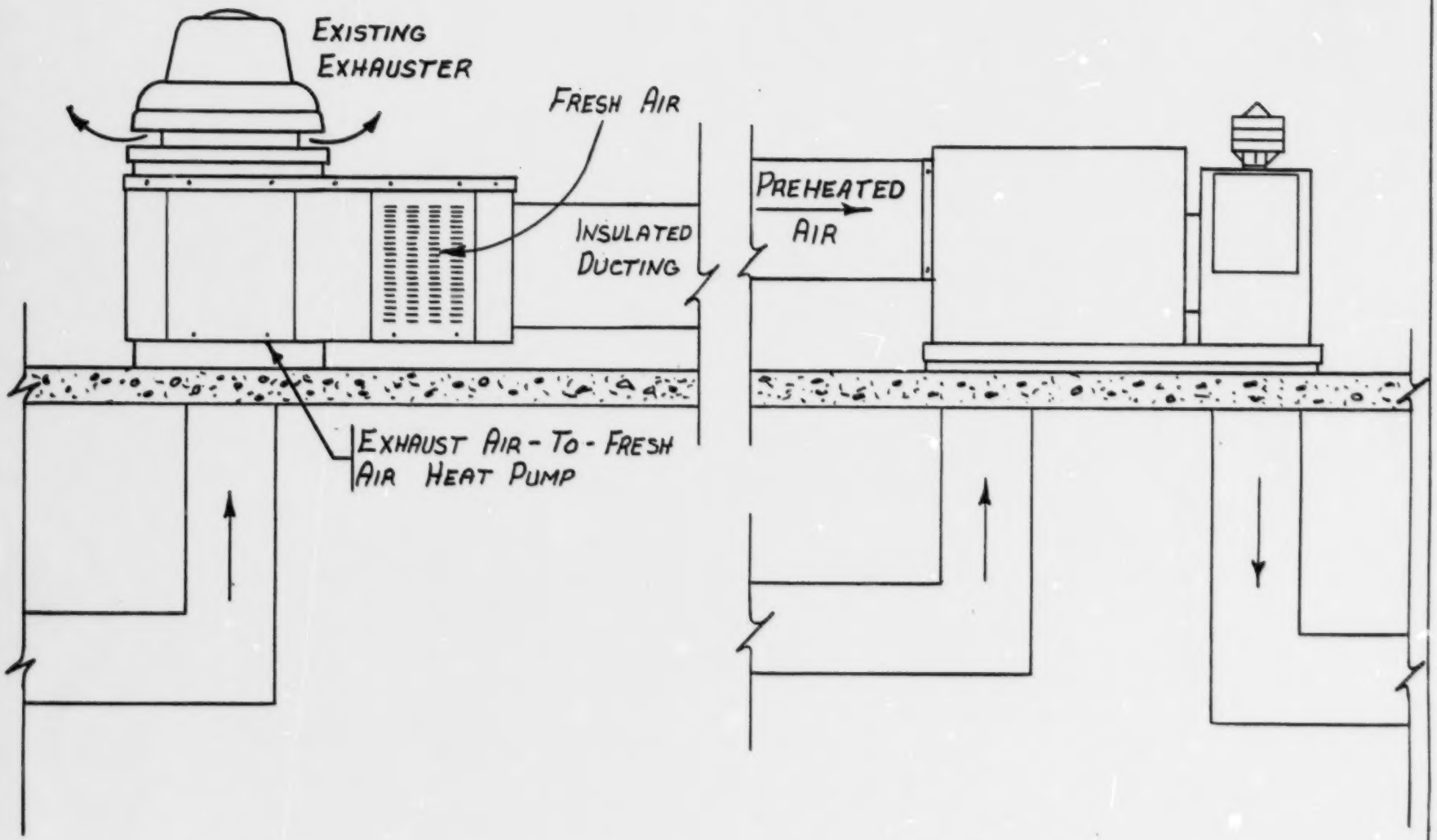
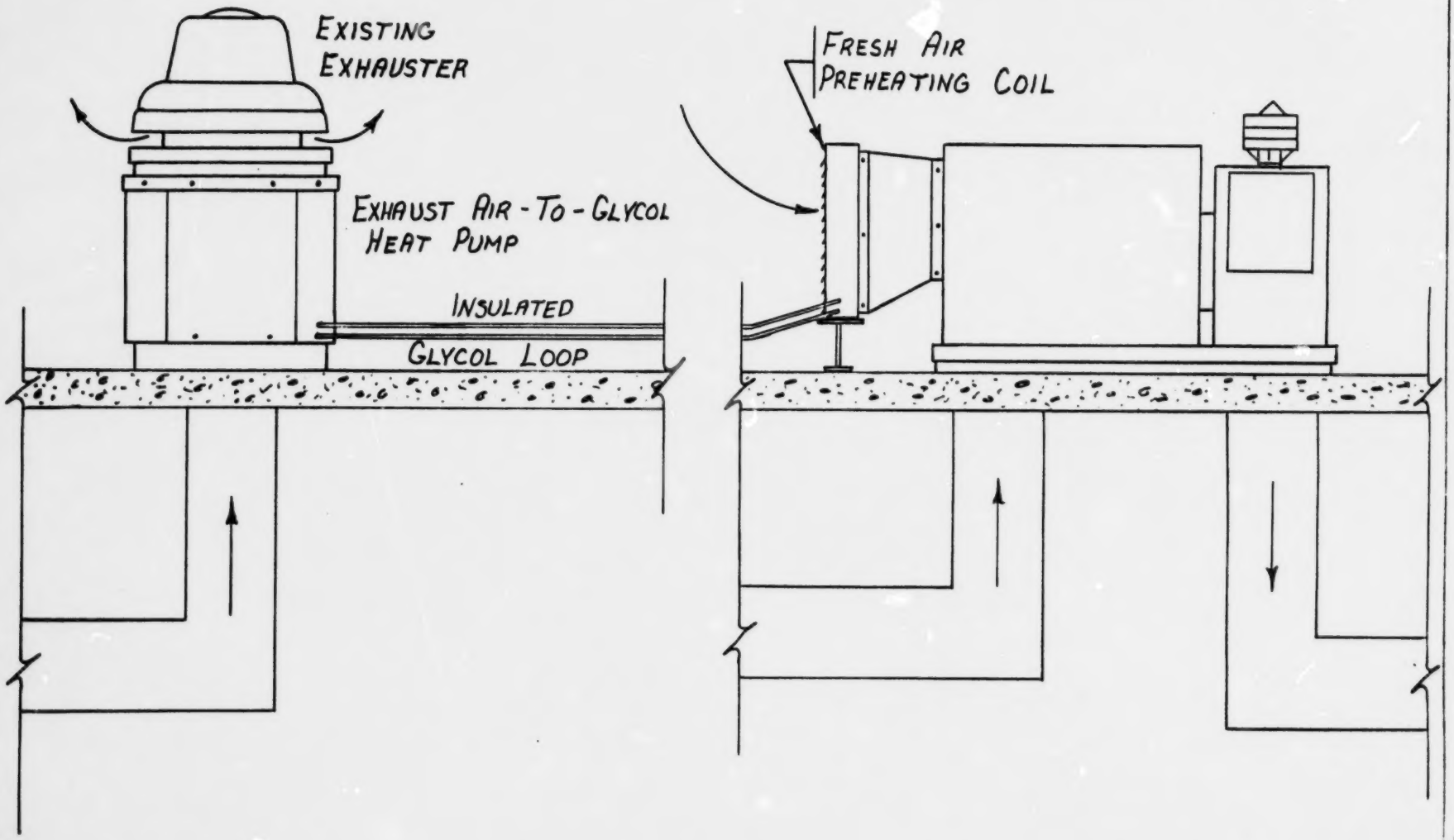


Fig. 2.5

Application Of An Exhaust Air-To-Fresh Air Heat Pump With Intermediate Glycol Loop



2.2.4 Exhaust Air-to-Supply Air Heat Exchanger

The air-to-air heat exchanger shown in Figure 2.6 is typical of the plate type heat exchangers as produced by Indusco. Because of the high pressure drop across the narrow air passages, supply and exhaust fans are included with unit. A bypass duct would be required so that the rooftop unit could provide free cooling to the building.

2.2.5 Glycol Run-Around Loop

In this system a glycol mixture would be circulated between a coil in the exhaust air duct and a coil in the fresh air intake (Figure 2.7). Because of the air flow blockage of the coil tube and fins, the coil and connecting ductwork would have to be larger than the existing ductwork. During the cooling season the glycol pump would be shut off.

2.3 System Comparison

Other than the thermal and economic system performance discussed in Sections 3 and 4, there are other comparisons that can be made that effect the feasibility of the systems. Some of these factors are presented in Table 2.1. If a building owner needed to install an air conditioning system, he might be interested in the EA-SA-HP or the EA-G-SA-HP systems. Hospitals and other health care facilities may not use EA-SA-HP or ATA-HE because of cross-contamination concerns.

Fig. 2.6

Application Of An Air-To-Air Heat Exchanger

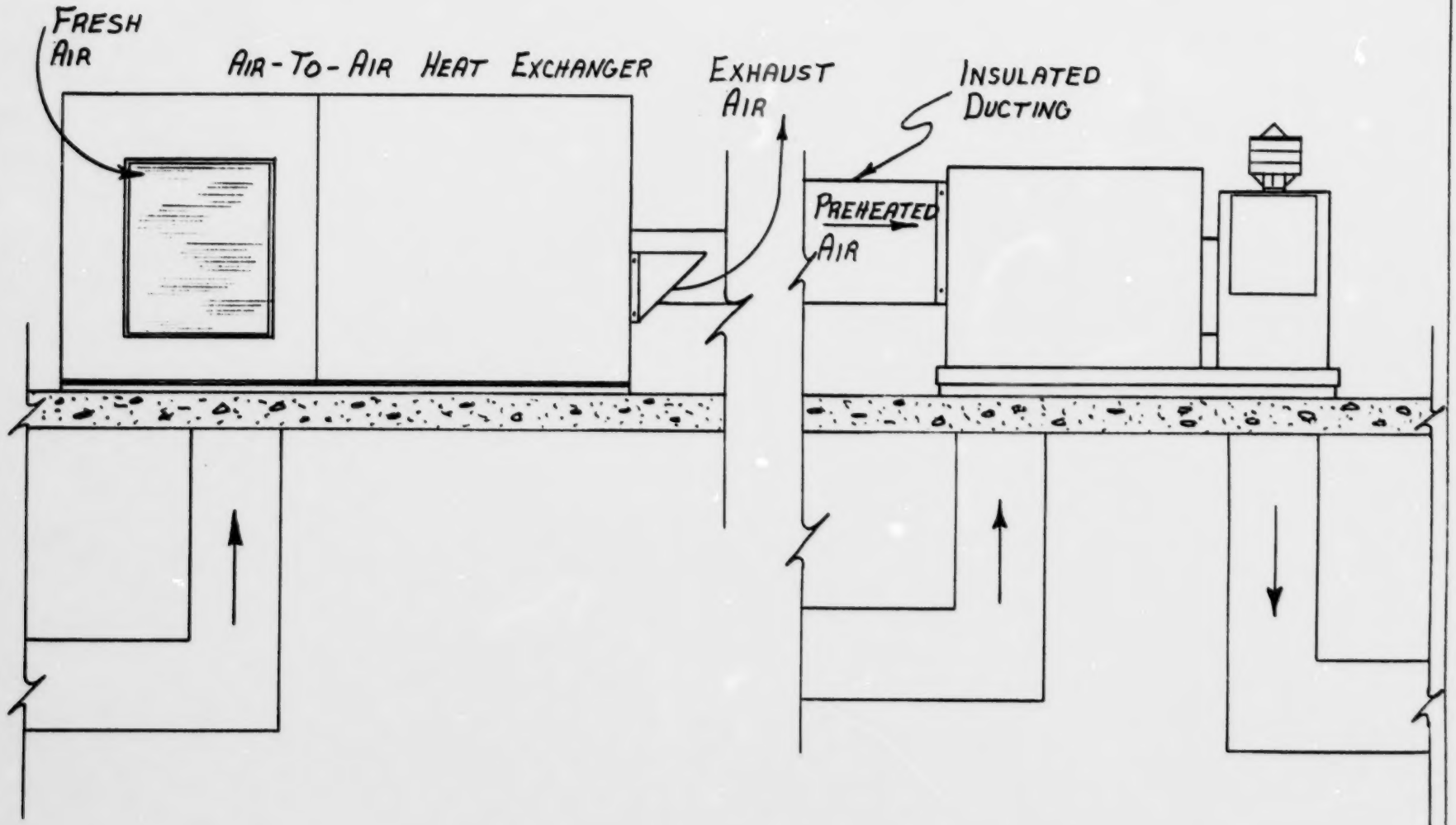


Fig. 2.7

Application Of A Glycol Run-Around Loop

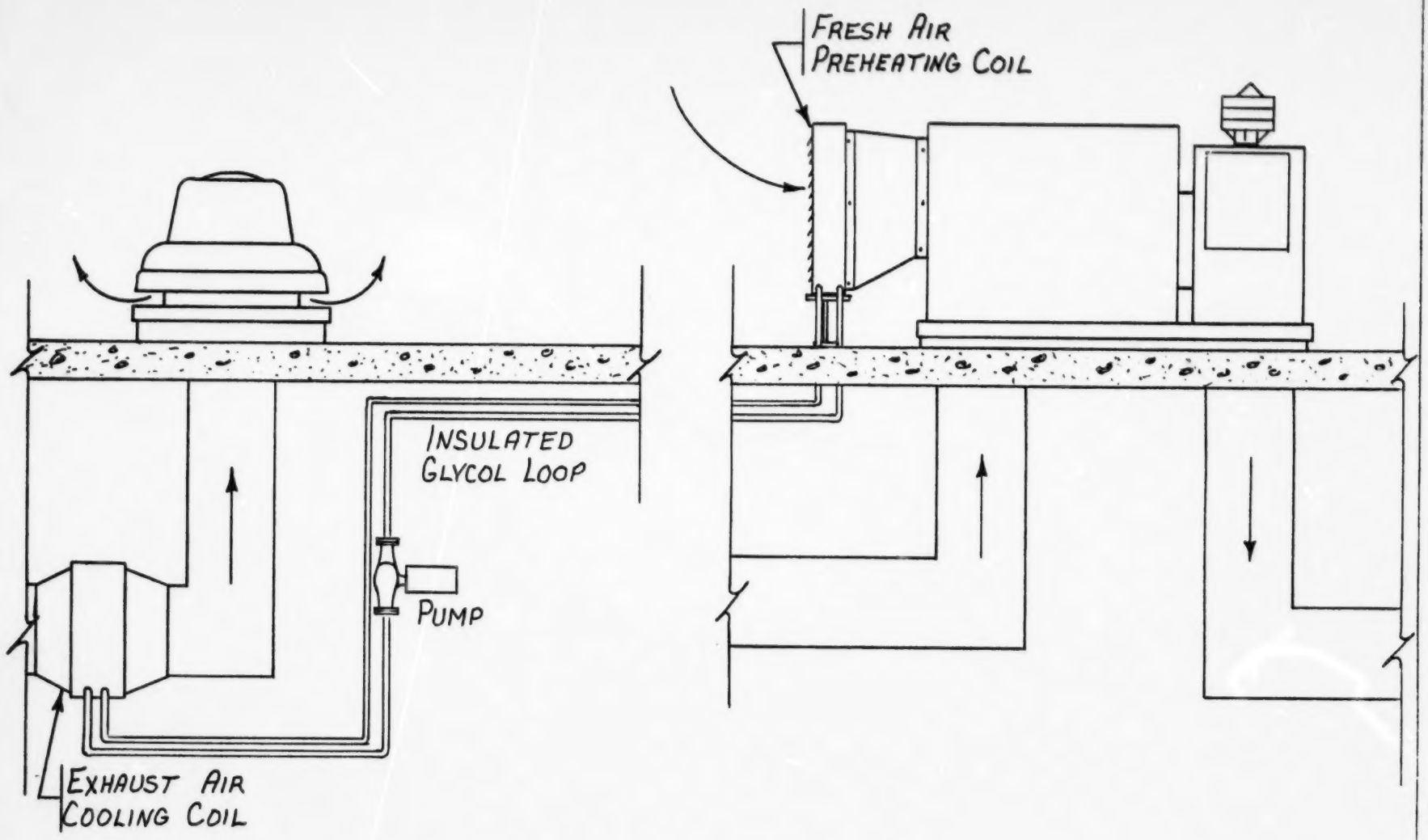


Table 2.1: System Comparison

Factor	EA-W-HP	EA-SA-HP	EA-G-SA-HP	EA-SA-HE	EA-G-SA
year round heat recovery	yes	no	no	no	no
air conditioning	no	yes	yes	no	no
on-site ductwork	no	yes	no	yes	no
water load required	yes	no	no	no	no
cross contamination potential	no	yes	no	yes	no
defrost required	no	not usually	not usually	yes	maybe
maintenance	little	little	some	little	some

3. SYSTEM PERFORMANCE

3.1 The Methodology

In order to assess the performance of the systems it is necessary to select design conditions and flow rates. This section will compare the energy and cost savings of the five systems exhausting air at 1000 l/s, 21°C and 40% R.H. While a flow rate of 1000 l/s may be viewed as low for commercial/industrial applications, it is, however, the most commonly used size of exhauster [4]. Because of economics of scale, installation of large flow rate heat recovery systems would be more cost effective. Thus the economic analysis in Section 4 may be viewed as a conservative assessment.

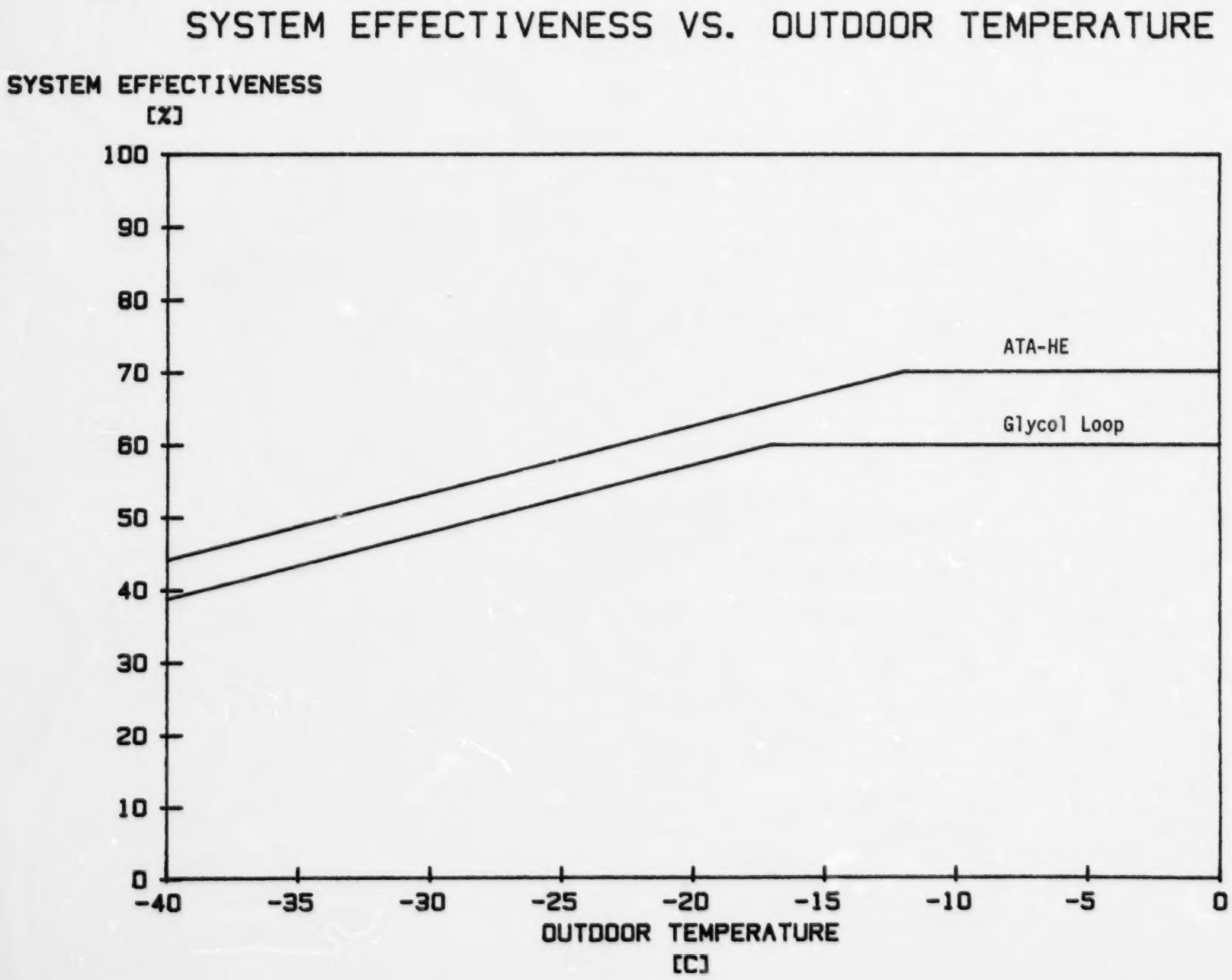
In calculating energy savings, no credit has been given for air-conditioning for two reasons. First, for all of the systems, savings in air-conditioning costs would be small. Second, many commercial/industrial buildings do not have air-conditioning.

3.1.1 Heat Exchanger Model

The performance of the two heat exchanger systems is better documented in the literature than the heat pump systems. Manufacturers data on air-to-air heat exchangers (plate type) give a nominal effectiveness of approximately 0.7 with balanced air flows [5]. A well designed glycol run-around loop should have an effectiveness of 0.6 [6].

Recent testing of residential air-to-air heat exchangers have shown a dramatic decrease in performance at cold outdoor temperatures [7]. The performance reduction is caused by frost build-up on the flow passages reducing heat transfer and shutting down the supply fan to allow the unit to defrost. If the supply fan is shut down, it has been assumed that an amount of air equal to the exhaust air flow rate must leak into the building (with no heat recovery). Performance curves for the two heat exchanger systems are shown in Figure 3.1. The decrease in effectiveness at cold temperatures was determined from testing done at Ontario Research Foundation [8] and as recommended by ASHRAE [6].

Fig. 3.1


SYSTEM EFFECTIVENESS VS. OUTDOOR TEMPERATURE
SYSTEM EFFECTIVENESS
[%]
100
90
80
70
60
50
40
30
20
10
0
ATA-HE
Glycol Loop
-40
-35
-30
-25
-20
-15
-10
-5
0
OUTDOOR TEMPERATURE
[C]

A factor often excluded in the analysis of heat recovery systems is parasitic power. That is the electrical energy required to overcome frictional losses in moving air and/or antifreeze. All five systems use some parasitic power, but it is especially important with air-to-air heat exchangers. In order to achieve high effectiveness, the air flow passages are made small resulting in a high pressure drop. For all systems parasitic power was included in the analysis.

3.1.2 Heat Pump Model

Because the heat pump designs under study are unique, there are few in-situ performance results available. The performance of the heat pumps were estimated using a computerized version of a National Research Council heat pump design tool [9]. Through an iterative procedure the method uses curve-fits to compressor, evaporator and condenser heat transfer and user-specified operating conditions to estimate system performance. Provided the system is properly designed and the program stays within the range of the curve-fits, the method should provide reasonably accurate results. The method was developed to simulate performance of air-to-air heat pumps. Enhancements and modifications, however, were made to the method to permit modelling of the two other heat pump systems under study. A complete description of the program algorithm is given in Appendix A.

The thermal performance of a commercially sized EA-W-HP has been tested and compared to the NRC method [10]. The laboratory results are within 10% of the method predictions for heat delivered. It is expected that the predictions of the other systems would be of similar accuracy.

As the outdoor temperature drops so does the temperature difference between the evaporator and condenser. At very cold outdoor temperatures it is possible for the refrigerant temperature difference and therefore, the pressure difference between the condenser and evaporator to become very low. A minimum of 170 kPa (25 psi) pressure drop is needed across the thermostatic expansion valve for it to function properly [11].

When typical pressure drops across other components are included, the minimum allowable pressure difference between condenser and evaporator is 240 kPa (35 psi). It has been assumed that some method of maintaining this pressure difference has been included in the system (e.g. refrigerant bypass). This restriction resulted in the EA-SA-HP and EA-G-SA-HP performance being held constant at outdoor temperature below -16°C and -26°C respectively.

By proper sizing of the heat pump components, it is possible to avoid frosting of the evaporator coil. The prediction method showed that the exhaust air temperature never went below 8°C and the evaporator temperature was always above -1°C. Thus, the heat pump model did not include any provision for decreased performance because of defrosting requirements.

Heat delivered and Coefficient of Performance (COP) curves as predicted by the model for the three systems are shown in Figures 3.2 and 3.3. Unlike most other heat pump systems, two of the systems have better performance the colder the outdoor temperature. The EA-W-HP performance is, of course, independent of temperature.

3.1.3 Annual Performance Calculation Method

For all the systems, the annual energy savings were estimated using the "bin" technique. The annual performance is the sum of the performance at each "bin" temperature multiplied by the number of hours in a year that each bin temperature occurs and the fraction of the time the system operates. The system may not operate for three reasons: exhaust fan is shut off, there is no heating load or there is a defrost requirement. The effect of defrost was discussed in the previous sections. The effect of non-continuous exhauster operation on system cost effectiveness is discussed in Section 4. The heating load requirement for the exhaust air to water heat pump system could be as long as 8760 hours (365 days x 24 hours) in a year since water heating is a year round load. For the other systems, however, the total number of hours that each system would

Fig.3.2

HEAT PUMP ENERGY DELIVERED VS. OUTDOOR TEMPERATURE

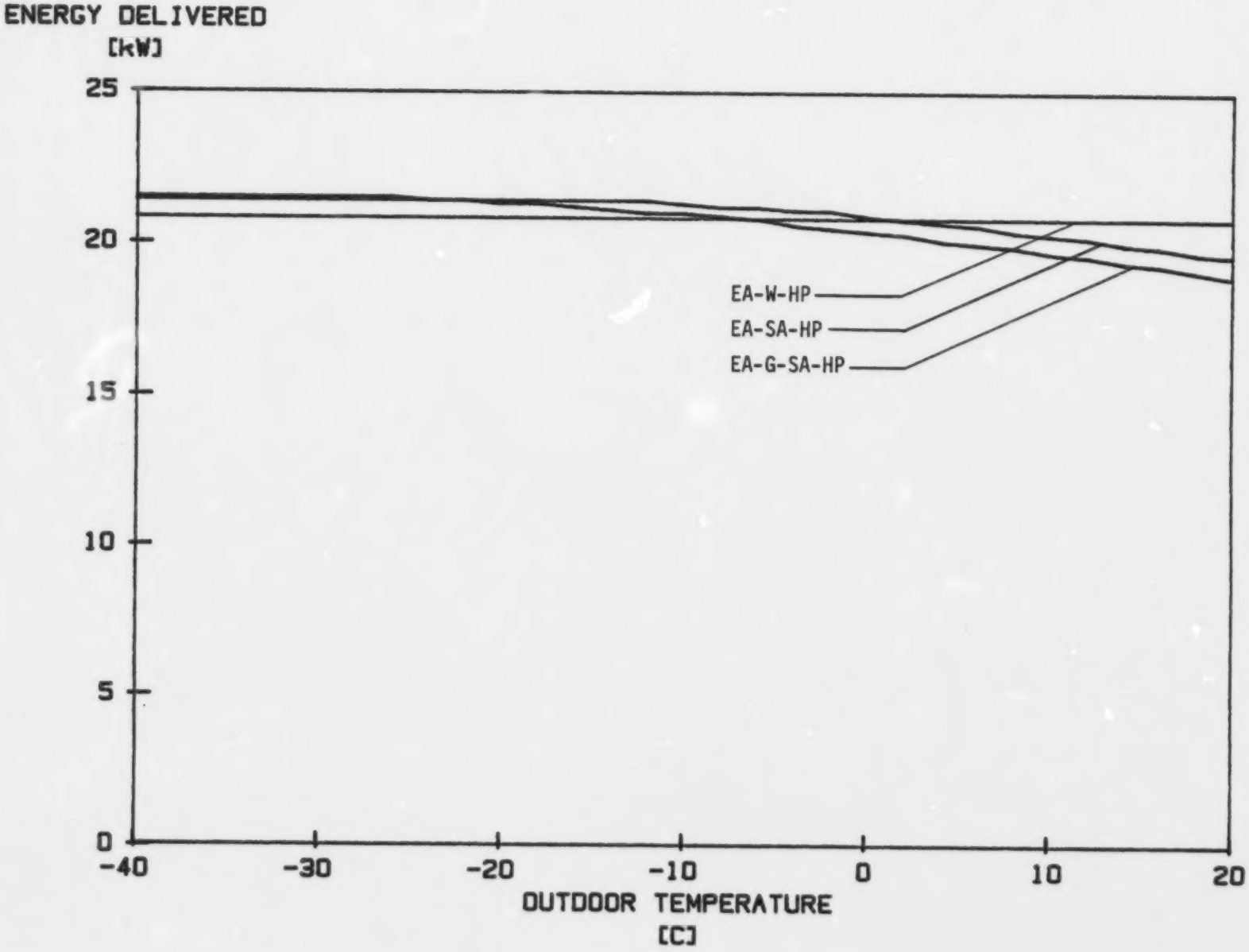
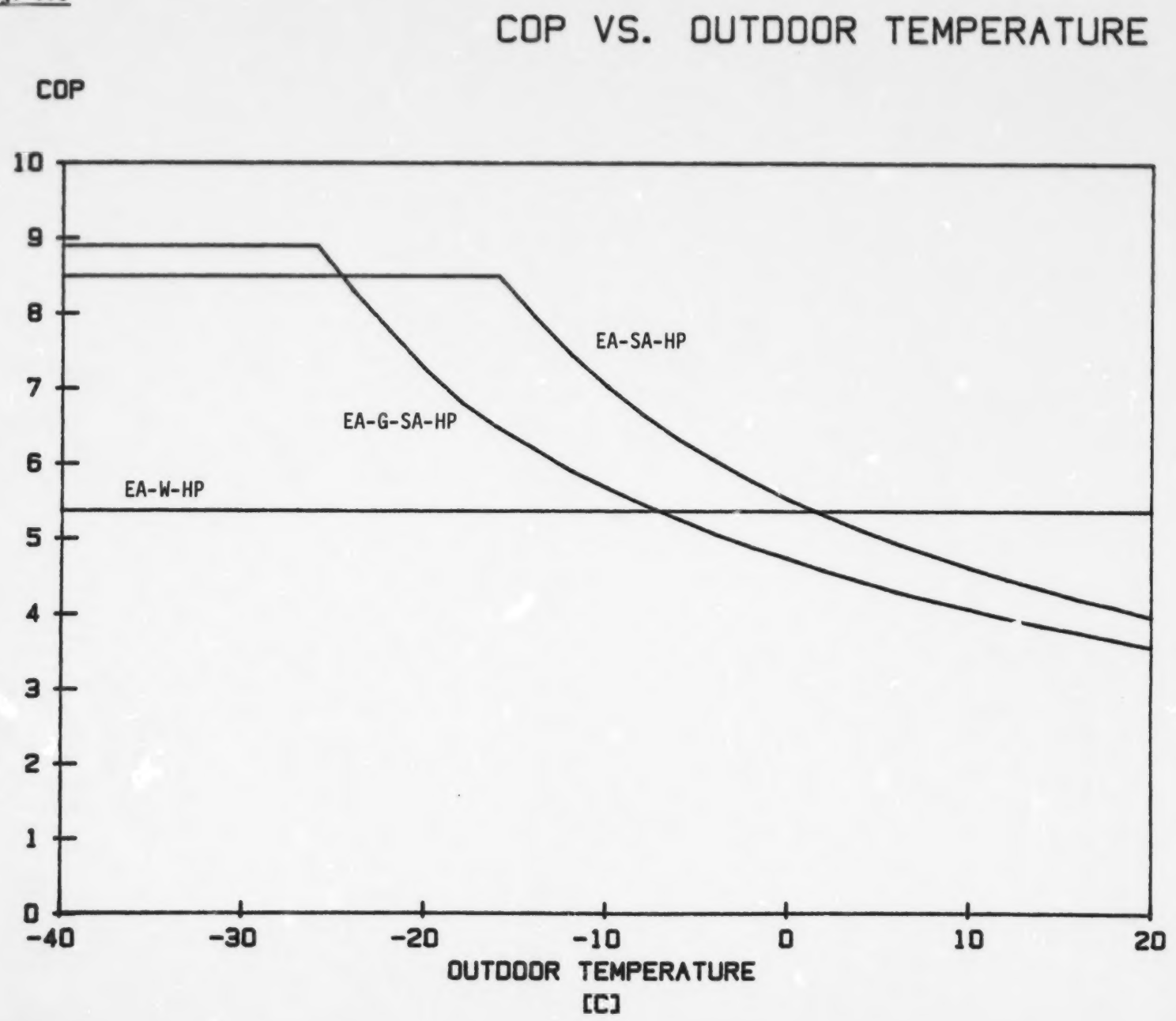


Fig. 3.3


COP VS. OUTDOOR TEMPERATURE
COP
10
9
8
7
6
5
4
3
2
1
0
EA-SA-HP
EA-G-SA-HP
EA-W-HP
-40
-30
-20
-10
0
10
20
OUTDOOR TEMPERATURE
[C]

operate is limited to those hours when the building requires space heating. The number of hours that the building requires heating is a function of its geographic location and its thermal characteristics. While degree-days, based on 18°C, are normally used to calculate heating requirements, most commercial/industrial buildings do not require heat unless the outdoor temperature is lower. For this study it has been assumed that the building does not require heating if the outdoor temperature is above 15°C.

Since on mild days the heat pump can deliver supply air warmer than the building temperature, it is assumed that this extra heat can be used to offset space heating costs. For this purpose the total building heating load was assumed to be three times the ventilation heating load.

The cost savings for each system were calculated using the formula:

Cost Savings = Energy Supplied x Cost of Heating Fuel/Seasonal Furnace Efficiency - Compressor and/or Fan Energy x Cost of Electricity

It has been assumed that the heating fuel is natural gas except for the Canadian Maritimes where #2 fuel oil is used. The seasonal efficiency of the heating system varies considerably from site to site. While the instantaneous combustion efficiency is generally close to 80%, factors such as heater warm-up, oversizing, and improper settings reduce the efficiency significantly. For this study an average value of 65% has been used.

To assess the feasibility of the heat recovery systems, the cost savings were examined in 11 locations across Canada and the northern U.S. The annual degree-days, electricity cost and natural gas cost for each of these locations are given in Table 3.1. These locations were chosen so as to cover the range of electricity and natural gas costs encountered in the two countries. There is a factor of over four between high and low electricity costs and over three between high and low natural

Table 3.1 Geographic Locations Studied

City	Degree Days (°C) (15°C Base)	Incremental Electricity Cost[1] (¢/kWhr)	Incremental Natural Gas Cost[1] (¢/m³)	Cost[2] Ratio (Elec/Nat. Gas)
Omaha, Nebraska	3673	3.47	20.2	1.78
Halifax, N. S.	4104	5.21	28.3[3]	1.97
Montreal, Quebec	4437	3.75	17.7	2.17
Winnipeg, Man.	5864	2.85	12.7	2.31
Toronto, Ont.	3658	4.47	15.5	2.98
Vancouver, B. C.	3031	3.73	12.7	3.03
Madison, Wisconsin	4368	7.23	24.2	3.09
Detroit, Michigan	3462	11.26	24.7	4.71
Boston, Mass.	3130	12.60	23.9	5.45
Edmonton, Alberta	5600	4.45	07.7	6.01
New York, New York	2706	15.18	24.1	6.51

1 in Canadian cents

2 Cost of Electricity in ¢/kWhr/(Cost of Natural Gas in ¢/m³ x 10.34 kWhr/m³), operating efficiency not included

3 Cost is in ¢/litre of #2 fuel oil

gas costs. The cities are listed in order of increasing ratio of electricity to natural gas cost. This is an important ratio since heat pumps use electricity to displace natural gas.

Obtaining representative electricity and natural gas costs is very difficult. Both of these fuels, but especially electricity, is on a sliding price scale, that is the more you use the cheaper the next unit of energy is. With energy conservation devices, the cost savings should be calculated at the incremental energy cost, not the average cost. Unfortunately the incremental cost varies with usage and usage varies with building type and size and time of year. The method used to calculate the incremental cost of energy should be reasonable for an "average" commercial/industrial building. The reader is cautioned, however, that when assessing the cost savings of a particular building, its incremental cost should be used. The Canadian and U.S. governments do not publish incremental energy costs, but rather fuel bills for various levels of consumption and demand. The following formula was used to estimate the incremental electricity cost using prices given in references 12 and 13:

$$\text{Incremental Electricity Cost} = \frac{\text{High Consumption Fuel Bill} - \text{Low Consumption Fuel Bill}}{\text{High Consumption} - \text{Low Consumption}}$$

For Canadian and U.S. locations the high and low monthly electricity consumptions were taken as 400000 kWhr and 10000 kWhr. The U.S. prices have been converted to equivalent Canadian funds.

It is important to note that this formula when used for electricity will result in demand charges being factored into the incremental energy cost. Since operating the heat pump will probably increase the demand charge as well as the consumption charge, the incremental energy cost is a simple means of incorporating both these changes.

For natural gas, consumption figures were not given in references 12 and 14. The incremental gas cost was taken to be the average industrial cost.

3.2 Thermal Performance of the Heat Recovery Systems

The rate of energy savings as a function of outdoor temperature for each of the five systems sized for a 1000 l/s exhaust air rate is shown in Figure 3.4. Energy savings rate is the heat output minus heat input (including parasitic power). The reader is cautioned that a graph of cost savings would look very different for locations where there is a large price difference between natural gas and electricity. The graph does, however, reveal that the thermal performances of the heat pump systems are relatively insensitive to outdoor temperature. (The exhaust air-to-water heat pump is, of course, independent of outdoor temperature and is represented by a straight line.) This is in direct contrast to outdoor air-source heat pumps where performance drops off significantly with decreasing outdoor air temperature. In fact at very cold temperatures outdoor air-source heat pumps would have an energy savings rate only slightly above zero (i.e. a COP slightly greater than one).

The performances of the air-to-air heat exchanger and glycol run-around loop systems are directly proportional to outdoor temperature down to the defrost temperature. Below the defrost temperature, the energy savings rate is almost constant.

The annual energy savings can be found by weighting the performance at each outdoor temperature by the number of hours that the temperature occurs. Table 3.2 gives the annual energy savings for the five systems in the 11 locations. The exhaust air-to-water heat pump saves the most energy because it operates all year. The two heat exchanger systems are much more sensitive to location than are the heat pump systems. Whereas the heat exchangers save more energy in a cold location (e.g. Edmonton), the heat pumps save more energy in warmer locations (e.g. New York).

Fig. 3.4

ENERGY SAVINGS RATE VS. OUTDOOR TEMPERATURE

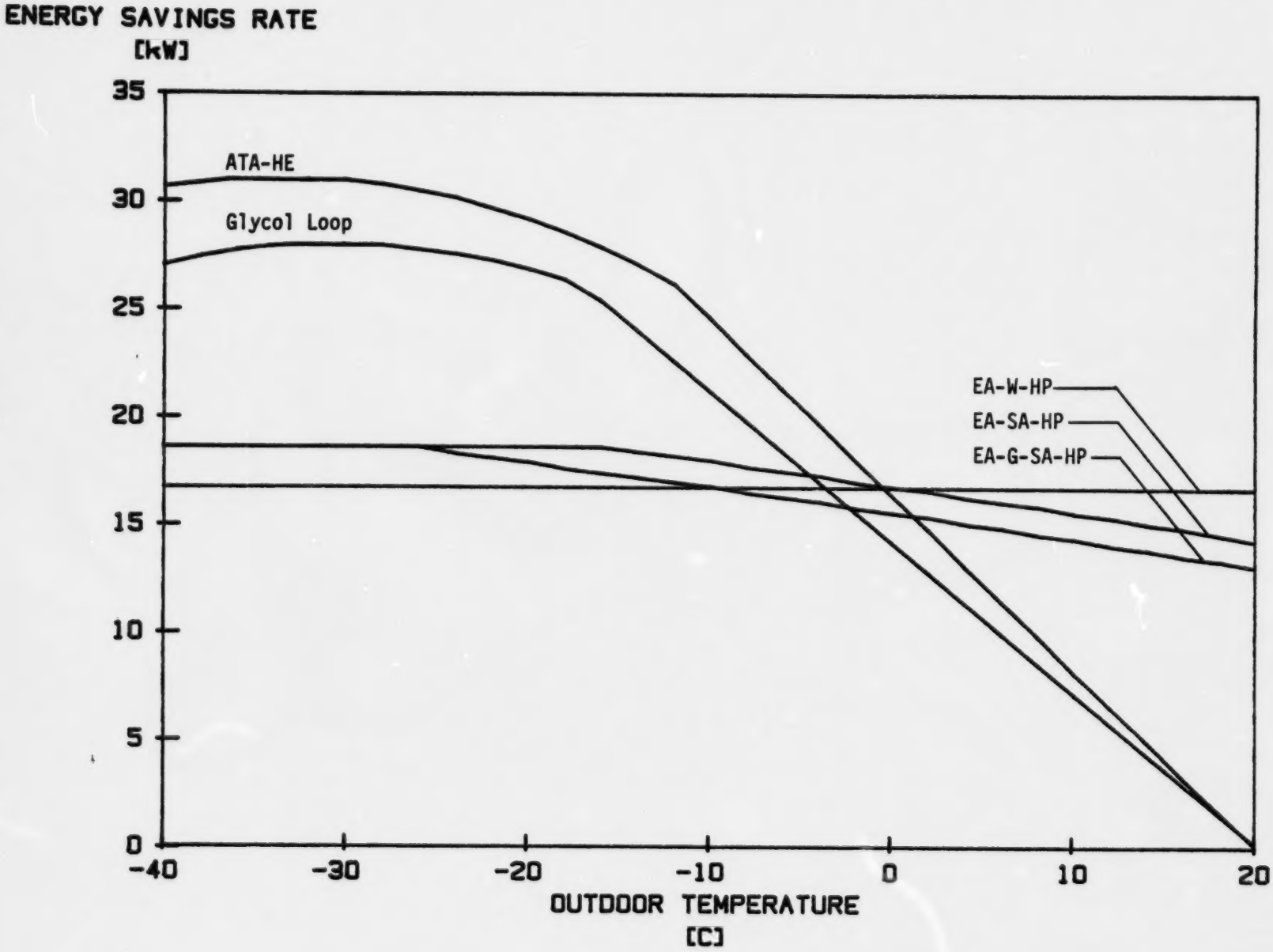


Table 3.2 Annual Energy Savings (in MWhr) [1]

City	EA-W HP	EA-SA HP	EA-G-SA HP	ATA-HE	GLYCOL LOOP
Omaha	146	76	70	74	65
Halifax	146	104	97	100	86
Montreal	146	97	90	99	86
Winnipeg	146	107	100	125	111
Toronto	146	94	86	96	81
Vancouver	146	102	95	82	72
Madison	146	88	82	90	79
Detroit	146	83	77	81	71
Boston	146	85	79	76	66
Edmonton	146	116	109	131	116
New York	146	75	70	65	59

1 Listed in order of increasing electricity to natural gas cost ratio.

3.3 Cost Savings of the Five Systems

In order to get a true assessment of the performance of each of the systems, the thermal performances must be multiplied by the fuel costs. Table 3.3 gives the annual cost savings (in Canadian dollars) for systems in each of the 11 locations The number in brackets after the cost savings is the rank of each system (in cost savings) for that location. For example in Halifax, the exhaust air-to-water heat pump saves the most money and the glycol run-around loop the least amount.

The exhaust air-to-water heat pump saves the most amount of money for eight of the locations because it can operate all year. In general locations with a low electricity to natural gas cost ratio save more money than locations with a high ratio regardless of location. It would appear that Edmonton and New York are particularly difficult locations to install any heat recovery system. In fact the EA-G-SA-HP has a negative savings in New York.

Table 3.3 Annual Cost Savings (in Canadian dollars) [1]

City	EA-W HP	EA-SA HP	EA-G-SA HP	ATA-HE	GLYCOL LOOP
Omaha	4224 (1)	2188 (3)	2016 (4)	2204 (2)	1952 (5)
Halifax	5519 (1)	3928 (3)	3581 (4)	3951 (2)	3436 (5)
Montreal	3444 (1)	2289 (3)	2072 (5)	2492 (2)	2193 (4)
Winnipeg	2414 (1)	1794 (4)	1636 (5)	2260 (2)	2060 (3)
Toronto	2587 (1)	1666 (4)	1348 (5)	2003 (2)	1749 (3)
Vancouver	2095 (1)	1421 (2)	1201 (5)	1362 (3)	1298 (4)
Madison	3948 (1)	2405 (4)	2046 (5)	2945 (2)	2742 (3)
Detroit	2626 (1)	1504 (4)	1009 (5)	2371 (3)	2407 (2)
Boston	1924 (3)	1038 (4)	480 (5)	1929 (2)	2099 (1)
Edmonton	479 (4)	479 (3)	240 (5)	1144 (2)	1209 (1)
New York	1045 (3)	418 (4)	-164 (5)	1459 (2)	1812 (1)

RANKING TOTALS

1st	8	0	0	0	3
2nd	0	1	0	9	1
3rd	2	4	0	2	3
4th	1	6	2	0	2
5th	0	0	9	0	2

1 Listed in order of increasing electricity to natural gas cost ratio.

4. SYSTEM COSTING AND COST EFFECTIVENESS

4.1 System Costs

The estimated installed cost of each system is given in Tables 4.1 to 4.5. The heat pump costs wer estimated by taking wholesale component prices and contractor estimated fabrication and assembly prices and scaling them up by a factor of three. A detailed breakdown of heat pump equipment and costs is given in Appendix B. The three times factor is to cover marketing, selling, profit and overhead costs. All installation prices were estimated by a Waterloo mechanical contractor. For the purposes of this study it has been assumed that regional variations in contracting costs are small. No frieght charges have been included since at present there is no fabrication location. All contractor prices include taxes and mark-up/profit.

Young and Lange [15] have estimated the installed cost of air-to-air residential heat pumps (converted to 1986 dollars) as $1200 plus $1200 per ton. Thus the five ton air-to-air unit under study should have an installed price of approximately $7200. The estimated installed price of $8320 is consistent with Young's findings when it is realized that the difference can be attributed to the extra ductwork requirement ($1400).

The prices of the glycol run-around loop coils and air-to-air heat exchanger were obtained from manufacturers/dealers of these products (Aerofin and McCarthy and Robinson respectively). A system design and commissioning charge has been included in the glycol run-around system. At present there is no company marketing a pre-designed packaged glycol run-around system. Thus, unlike the other systems, an engineering firm or contractor would have to size and select the pump and coils. The system would also have to be inspected/commissioned upon completion to ensure proper operation. While the same argument could be applied to the exhaust air-glycol-supply air heat pump, it has been assumed that the manufacturers three times mark-up would include preparation of design guides and design assistance to the dealer/installer.

Table 4.1 Installed Cost of EA-W-HP

Components	Cost
1) EA-W-HP (Including taxes)	$5830.
2) Install EA-W-HP	330.
3) Supply and install piping (120' - 3/4")	2200.
4) Electrical	400.
5) System Rebalance	120.
	$8880.

Table 4.2 Installed Cost of EA-SA-HP

Components	Cost
1) EA-SA-HP (Including taxes)	$6000.
2) Install EA-SA-HP	400.
3) Supply and Install Ductwork (35')	1400.
4) Electrical	400.
5) System Rebalance	120.
	$8320.

Table 4.3 Installed Cost of EA-G-SA-HP

Components	Cost
1) EA-W-HP (Including taxes)	$5830.
2) Install EA-W-HP	330.
3) Air Coil	1130.
4) Supply and Install piping (70' - 1")	1710.
5) Electrical	650.
6) System Rebalance	120.
	$10420.

Table 4.4 Installed Cost of ATA-HE

Components	Cost
1) ATA-HE	$7000.
2) Install ATA-HE with new curb	760.
3) Supply and Install ductwork (28')	1100.
4) Electrical	250.
5) System Rebalance	75.
	$9185.

Table 4.5 Installed Cost of Glycol Run-around Loop

Component	Cost
1) 2 Air Coils	$2370.
2) Install Coils and Transitions	350.
3) Pump	500.
4) Supply and Install Piping (100' - 1½")	2200.
5) Electrical	250.
6) System Rebalance	220.
7) System Design and Commissioning	1500.
	$7390.

It was assumed that the conventional water heater was 15 metres from the roof exhauster and the supply and exhaust ducts were 10 metres apart. In comparing the installation costs given to what it would be for other buildings, it should be noted that piping and ducting costs do not increase or decrease linearly. If longer piping or duct runs were needed, these costs would increase only slightly since much of the cost is in setting-up and making the connections to existing piping or ductwork.

4.2 Cost Effectiveness

In assessing the cost effectiveness of the systems many different techniques can be used. Techniques such as net present value and discounted payback take into account such factors as fuel escalation rate, lending rates, tax rate, maintenance costs and incentive programs. The reality of the situation is that in most cases the purchaser will probably make his decision based on simple payback. The other factors will alter what simple payback is acceptable to the purchaser. Governments and non-profit institutions might accept a maximum of a five year simple payback, whereas, industries might require a two year simple payback before purchase. This report will use simple payback to compare the systems. It is left to the reader to determine whether the calculated simple payback is acceptable to their building sector.

Dividing the system costs given in Section 4.1 by the cost savings of Section 3.3 gives the simple payback in years. Table 4.6 gives the simple payback for the systems in the 11 locations. The number in brackets after the payback is the rank of that system for that locaton. The exhaust air-to-water heat pump and glycol run-around loop are the most cost effective systems. The EA-W-HP system gives the best payback in six of the locations and the glycol loop the remaining five. For locations other than those with high electricity to natural gas cost ratios the exhaust air-to-supply air heat pump and heat exchanger offer somewhat similar paybacks. Thus, the heat pump has a slightly better performance in locations with low electricity costs, whereas the heat exchanger would be favoured where electricity costs are high. The EA-G-SA-HP although being a versatile system has the longest payback period; generally over five years. This payback period is probably unacceptable to most building owners.

Table 4.6 Simple Payback (in years) [1]

City	EA-W HP	EA-SA HP	EA-G-SA HP	ATA-HE	GLYCOL LOOP
Omaha	2.1 (1)	3.8 (3)	5.2 (5)	4.2 (4)	3.8 (2)
Halifax	1.6 (1)	2.1 (2)	2.9 (5)	2.3 (4)	2.2 (3)
Montreal	2.6 (1)	3.6 (3)	5.0 (5)	3.7 (4)	3.4 (2)
Winnipeg	3.7 (2)	4.6 (4)	6.3 (5)	4.1 (3)	3.6 (1)
Toronto	3.4 (1)	5.0 (4)	7.7 (5)	4.6 (3)	4.2 (2)
Vancouver	4.2 (1)	5.9 (3)	8.7 (5)	6.7 (4)	5.7 (2)
Madison	2.3 (1)	3.5 (4)	5.1 (5)	3.1 (3)	2.7 (2)
Detroit	3.4 (2)	5.5 (4)	10.3 (5)	3.9 (3)	3.1 (1)
Boston	4.6 (2)	8.0 (4)	21.7 (5)	4.8 (3)	3.5 (1)
Edmonton	18.6 (4)	17.4 (3)	43.5 (5)	8.0 (2)	6.1 (1)
New York	8.5 (3)	19.9 (4)	N/A (5)	6.3 (2)	4.1 (1)

RANKING TOTAL

1st	6	0	0	0	5
2nd	3	1	0	2	5
3rd	1	4	0	5	1
4th	1	6	0	4	0
5th	0	0	11	0	0

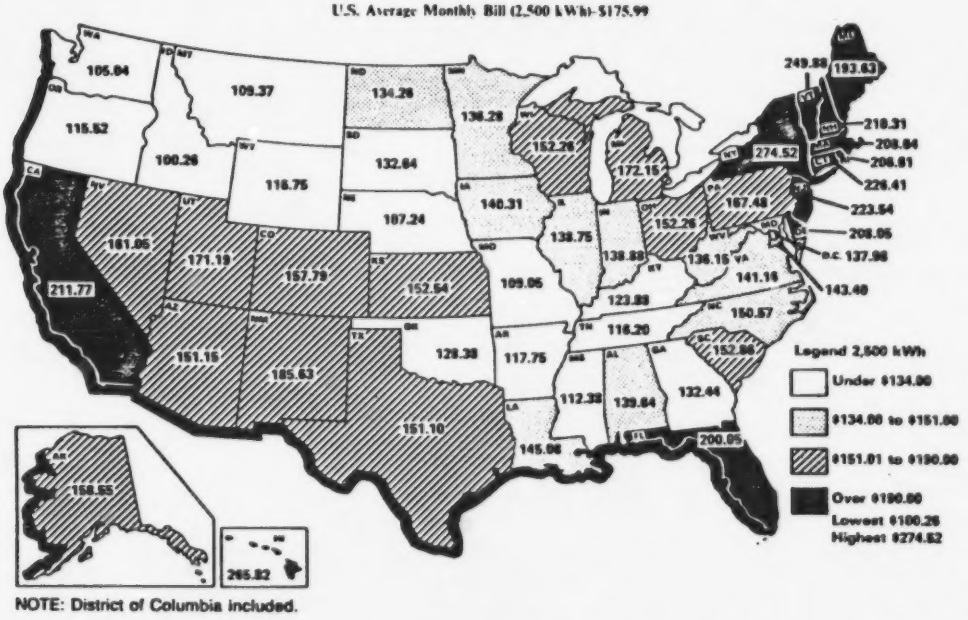
1 Listed in order of increasing electricity to natural gas cost ratio.

It is interesting to note the wide variation in a system's payback time depending on location, for example, the exhaust air-to-water heat pump payback period can vary from 1.6 years to over eighteen years. Clearly the potential of the heat pump system depends on the cost of energy and in particular the cost of electricity. In general, exhaust air heat pump systems have potential where the cost ratio of electricity to natural gas is less than five. Figure 4.1 shows a map of U.S. electricity costs. The heat pump systems should be most cost effective in the states with low electricity costs.

The payback times given in Table 4.6 assume continuous exhaust (i.e. 24 hours/day). To a very close approximation the payback period can be adjusted for shutdown periods by dividing by the fraction of the day that air is exhausted. For example, if air is exhausted only 12 hours a day, the payback period would be twice as long. It can be seen that if the exhauster is not operated at least 80 hours a week then even the best payback times would not be acceptable to most building owners.

Figure 4.1

Figure 6. State Average Monthly Bills, 2,500 Kilowatthour Residential, January 1, 1985.



NOTE: District of Columbia included.

Source: Energy Information Administration Form EIA-213, "Typical Net Monthly Bills."

Typical Electric Bills January 1, 1985
Energy Information Administration

5. MARKET ASSESSMENT

5.1 Market Size

The purpose of this portion of the market study is to estimate the potential market size for the five types of heat recovery units under consideration. All these units are commercially/industrially sized for installation on a nominal 1000 l/s or 2000 CFM exhaust system.

The technique to determine market size was to first establish the building categories which could potentially utilize these heat recovery systems, the number of buildings in each category, their floor areas, and any other related information regarding size. The second step was to estimate the total exhaust rate for each building type and normalize this figure in terms of the number of 2000 CFM exhaust systems required to meet this total exhaust rate. (Actual exhaust systems, of course, would cover a full range of flow rates from 50 CFM to over 20000 CFM.) This number of exhaust systems represents an absolute maximum of possible heat recovery unit locations if all ventilation exhaust was achieved through central ducting and multiple 2000 CFM units were used for large exhaust systems. The operating and feasibility requirements of the heat recovery units would reduce this number significantly. For each building category the fraction of buildings that met each of the four criteria were estimated. These criteria were:

1) Extended Hours: operates a minimum of 80 hours per week for 7 months of the year
2) Central Exhaust: ventilation exhaust occurs through central ducting and large (>1000 CFM) roof exhausters, rather than through multiple small (50-100 CFM) exhausters or natural infiltration
3) Water Demand: sufficient water heating demand (approx. 2000 U.S. gallons/day) to justify the use of the exhaust air-to-water heat pump. This water demand may be continuous flow or cycled through a storage tank.

4) Feasible Ducting: economically feasible to run insulated ducting between the existing exhaust and supply duct locations. This criteria would also reflect unadaptable situations such as building side horizontal exhausters and cross-contamination restrictions.

The potential market size for the exhaust air-to-water heat pump was the total market times the fraction of the market meeting criteria 1, 2 and 3. For the exhaust air-to-supply air heat pump and heat exchanger, the potential market is the total market times the fraction of the market meeting criteria 1, 2 and 4. The two glycol systems need only meet criteria 1 and 2.

The raw data required, as mentioned previously, was the number of buildings of each type, their floor areas, and any other related information regarding the building sizes. The major sources for this information were Statistics Canada publications and BETT (Building Energy Technology Transfer) Program publications, a Department of Energy, Mines and Resources Canada sponsored program.

The total Canadian market size of 2000 CFM heat recovery systems by building type is given in Table 5.1. The calculation of these numbers is given in Appendix C. The two glycol systems have the largest potential market at 165,000 units in Canada. The estimated Canadian potential market size for the exhaust air-to-water heat pump is 136,000 units with the exhaust air-to-supply air heat pump/heat exchanger market being significantly smaller at 102,000 units.

For all the systems the largest market is the habitation building category which includes apartment buildings, hospitals and special health care facilities. Restaurants and recreational facilities also represent a significant market for the products, although special precautions will be necessary to meet building code requirements for restaurant kitchen exhaust.

Table 5.1 Summary of the Canadian Market

Building Sector: Type	No. of Locations	Estimated No. of Exhausters	Exhaust Air to Water	Exhaust Air to Supply Air	Glycol Loop
HABITATION					
Apartment Buildings	78892 21.6%	64147 17.6%	16969 12.4%	13087 12.8%	18409 11.1%
Penitentiaries & Corre'l Fac.	240 0.1%	998 0.3%	593 0.4%	535 0.5%	714 0.4%
Special Care Facilities	5607 1.5%	23592 6.5%	18151 13.3%	8940 8.7%	19061 11.5%
Hospitals	1230 0.3%	26682 7.3%	25729 18.8%	5196 5.1%	26491 16.0%
Hotels	3948 1.1%	3476 1.0%	1195 0.9%	1108 1.1%	1500 0.9%
Motels	3759 1.0%	1090 0.3%	209 0.2%	77 0.1%	302 0.2%
SUBTOTALS	93676 25.7%	119984 33.0%	62846 46.0%	28943 28.3%	66477 40.2%
COMMERCIAL					
Retail Stores	86892 23.8%	13911 3.8%	47 0.0%	465 0.5%	931 0.6%
Restaurants	36668 10.1%	28906 7.9%	19769 14.5%	14952 14.6%	20353 12.3%
Motion Picture	1554 0.4%	2331 0.6%	0 0.0%	199 0.2%	221 0.1%
SUBTOTALS	125114 34.3%	45148 12.4%	19816 14.5%	15616 15.3%	21505 13.0%

INDUSTRIAL					
Office Buildings	1803	20104	334	2583	3446
	0.5%	5.5%	0.2%	2.5%	2.1%
Warehouses	52108	3488	4	15	110
	14.3%	1.0%	0.0%	0.0%	0.1%
Light Industrial	19135	44312	2294	3192	6304
	5.3%	12.2%	1.7%	3.1%	3.9%
Manufacturing Industry	37526	31584	2179	3330	6659
	10.3%	8.7%	1.6%	3.2%	4.0%
SUBTOTALS	110572	99488	4811	9120	16599
	30.3%	27.3%	3.5%	8.9%	10.0%
EDUCATIONAL					
Elem'y & Sec'y Schools	15519	22930	4552	3424	6848
	4.3%	6.3%	3.3%	3.3%	4.1%
Comm'y Colleges & Universities	260	1363	528	652	948
	0.1%	0.4%	0.4%	0.6%	0.6%
SUBTOTALS	15779	24293	5080	4076	7796
	4.3%	6.7%	3.7%	4.0%	4.7%
SERVICE					
Power Laundries	3397	5096	3439	2752	3439
	0.9%	1.4%	2.5%	2.7%	2.1%
Recreational Facilities	15815	70092	40611	41885	49690
	4.3%	19.3%	29.7%	40.9%	30.0%
SUBTOTALS	19212	75188	44050	44637	53129
	5.3%	20.6%	32.2%	43.6%	32.1%

TOTALS FOR CANADA

Number of Locations	364,353
Estimated Number of Exhausters	364,100
Exhaust Air-to-Water Heat Pump	136,603
Exhaust Air-to-Supply Air Heat Pump or Heat Exchanger	102,392
Exhaust Air/Glycol/Supply Air Heat Pump or Glycol Run-around Loop	165,506

6. CONCLUSIONS

As a result of this study a number of conclusions can be made about the use of exhaust air heat pumps.

- Exhaust air heat pumps have the advantages of no defrost requirement and the potential for providing air conditioning.

- EAHP's have seasonal coefficients of performance over five and instantaneous values as high as eight.

- In most locations the exhaust air-to-water heat pump offers the greatest energy and cost savings of the heat recovery systems studied.

- In general, in locations with a high cost ratio of electricity to natural gas (e.g. New England and the Prairies) the glycol run-around loop offers the shortest payback period; in locations with a low cost ratio of electricity to natural gas (e.g. central and eastern Canada and the U.S. mid-west) the exhaust air-to-water heat pump offers the shortest payback period.

- In all locations except Edmonton and New York the exhaust air-to-water heat pump achieved a simple payback of four years or less and an average payback of 2.8 years.

- Use of an intermediate glycol loop between the exhaust air heat pump and the supply air increased the payback period to beyond what most building owners would accept.

- In many locations the exhaust air-to-supply air heat pump offers similar paybacks to a conventional air-to-air heat exchanger and may be more advantageous if air conditioning is required.

- The market for exhaust air heat pumps (either to heat supply air or water) in Canada is very large. The estimated market size is 136,000 and 102,000 2000 CFM units for the water and air systems respectively. The majority of the market is in apartments, hospitals, restaurants and recreational facilities.

- It would appear that the exhaust air-to-water heat pump is a viable alternative to conventional heat recovery and its development and commercialization should be pursued.

7. REFERENCES

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Appendix A

HPUMP is a computer program, based on work done by the National Research Council, which uses manufacturer's published performance data and National Research Council test results to model heat pump system perfo:mance.

The inputs to the program are the compressor, evaporator, and condenser performance data, and the ambient conditions at the evaporator and condenser. The ambient conditions of interest are the entering air/water temperature to the condenser and the air velocity, temperature and relative humidity at the evaporator and condenser fin coils. The program will then calculate the required compressor work, the evaporator and condenser heat input/output, and the relevant system temperatures and air or water outlet conditions. The compressor, evaporator and compressor performance were each calculated using an equation which was the result of a curve-fitting method to the performance data.

The program then guesses an evaporating temperature (the saturation temperature of the refrigerant inside the evaporator) and uses an iterative approach to calculate the system parameters. Iteration stops when work into the system (i.e. compressor and evaporator) equals work out of the system (i.e. condenser and losses). Losses are assumed to be 5% of the evaporator work. Thus the program is complete when the residual is small.

residual = condenser work - 0.95 x evaporator work - compressor work

When modelling the EA-G-SA-HP, no glycol temperature is assumed, but rather the program guesses a glycol input temperature to the glycol/refrigerant heat exchanger and iterates until the glycol temperture into the glycol/refrigerant heat exchanger is the same as the glycol temperature coming from the glycol/air heat exchanger. This assumes that

pump work + condenser work = evaporator work

and assumes no system losses. The same curve fit equations were used for all three systems.

Appendix B: Heat Pump Production Costs

Part and Description	Estimated Production Costs	
	EA-W	EA-SA
Components		
Cabinet: Material	80.00	100.00
Labour	125.00	150.00
Painting	40.00	45.00
Compressor:	315.00	315.00
Condenser:	150.00	150.00
Evaporator:	150.00	150.00
Dual Pressure Control:	90.00	90.00
Transformer:	10.00	10.00
Contactor:	40.00	40.00
Accumulator:	35.00	35.00
Expansion Valve:	70.00	70.00
Distributor:	10.00	10.00
Sight Glass:	15.00	15.00
Filter/Drier:	15.00	15.00
System Controls: includes water temperature, air flow, water flow sensors and control elements	150.00	150.00
Back-draft damper and filter	150.00	150.00
Misc. small parts	35.00	35.00
Unit Assembly: includes installing major components and connecting refrigeration cycle; wiring controls, etc.	125.00	125.00
Refrigeration Test/Adjustment: test run unit and adjust controls accordingly; charge unit	60.00	60.00
Total Fabrication Cost	1665.00	1715.00
Overhead, Administration and Selling Costs (2 x Fabrication Cost)	3330.00	3430.00
Total Retail Price	4995.00	5145.00
Taxes	835.00	855.00
Total	$5830.00	$6000.00

APPENDIX C: EXHAUST AIR HEAT RECOVERY MARKET DETERMINATION

The purpose of this appendix is to estimate the potential market size for the five types of heat recovery units under consideration. All these units are commercially/industrially sized for installation on a nominal 2000 CFM exhaust system. Building size subdivisions used for this study correspond to those found in the original data sources. However, in some cases subdivisions were combined where it was thought such extensive size categories were not required.

As is common, not all data found was current. To bring all numbers to a common point (end of 1985), economic indicators closest related to the building category were used to obtain multiplying factors, or in cases where several previous years data was available, extrapolation was used. Table C.1 gives the economic indicators, the multiplication factors used and the references for each building type. Indicator factors were taken from Reference 22.

The final method of calculation for each size category and building type is outlined in Table C.2. Each spreadsheet line is discussed below:

(1) No. of Locations: The number of locations, as determined from the data, where this building type is found. In most cases one location refers to one building. However, in cases such as Universities or Penitentiaries, more than one building may exist at one location.

(2) 000's Sq. Ft.: The total floor area, as determined from the data, for the given building type.

(3) No. of [?]: The number of categorically related items which can be linked to the rate of ventilation for the buildings.

(4) Exhausters/Location: The estimated number of 2000 CFM exhausters which would be found at a single location. This value was only considered when other ventilation related data was available.

Table C.1: Summary of Raw Data

Building Sector: Type	Data References	Year of Data	Economic Indicator	Multiply Factor
HABITATION:				
Apartment Buildings	[1,19]	1981-82	Total + Starts('82,'83,'84)	
Penitentiaries	[9]	1983-84	Constant	1.000
Special Care	[3,14]	1985		
Hospitals	[2,15]	1985		
Hotels	[7]	1981-83	Extrapolation	
Motels	[8]	1981-83	Extrapolation	
COMMERCIAL:				
Retail stores	[6,16]	1983	Gross Dom. Prod.	1.0945
Restaurants	[17]	1978	Labour Force	1.1600
Motion Pic. Theatres	[4]	1982	Labour Force	1.0554
INDUSTRIAL:				
Warehouses	[12]	1981	Gross Dom. Prod.	1.0757
Office Buildings	[18]	1982	Labour Force	1.0554
Light Industrial	[13]	1981	Gross Ind'l Prod.	1.0755
Manufac's Industry	[10]	1983	Gross Dom. Prod.	1.0945
EDUCATIONAL:				
Elem'y & Sec'y Schools	[5]	1983-84	Constant	1.0000
Universities/Colleges	[5]	1983-84	Constant	1.0000
SERVICE:				
Power Laundries	[11]	1975	Labour Force	1.2671
Recreation Facilities	[20,21]	1972	Labour Force	1.3810

Table C.2: Method of Calculation

[1]	No. of Locations		#
[2]	000's Sq. Ft.		#
[3]	No. of (?)		#
[4]	Exhausters/Location		#
[5]	CFM Exh'd/1000 Sq. Ft.		#
[6]	CFM Exhausted/(?)		#
[7]	CFM Exhausted		[2] x [5]
		OR	[3] x [6]
[8]	No. of Exhausters		$\frac{[1] \times ([4] \text{ or } [7])}{2000 \text{ CFM}}$
[9]	Water Demand		#
[10]	Central Exhaust		#
[11]	Feasible Ducting		#
[12]	Extended Hours		#
[13]	Exh. Air to Water		[8] x [9] x [10] x [12]
[14]	E.A. to Supply Air		[8] x [10] x [11] x [12]
[15]	Glycol Loop		[8] x [10] x [12]

(5) CFM Exh'd/1000 Sq. Ft.: The recommended ventilation rate per 1000 sq. ft. of floor area as listed in Reference 23.

(6) CFM Exhausted/[?]: The recommended ventilation rate per related item, as listed in Reference 23.

(7) CFM Exhausted: The calculated total cu. ft. per min. ventilation rate for the given building category.

(8) No. of Exhausters: The equivalent number of 2000 CFM exhausters required to meet the total ventilation rate. This represents an absolute maximum of possible heat recovery unit locations.

(9) Water Demand: The estimated percentage of exhausters (from 8) with a sufficient water heating demand (approx. 2000 U.S. gallon/day) to justify the use of the exhaust air-to-water heat pump. This water demand may be continuous flow or cycled through a storage tank.

(10) Central Exhaust: The estimated percentage of ventilation exhaust which would occur through central ducting and large (> 1000 CFM) roof exhausters, rather than through multiple small (50 - 100 CFM) exhausters or natural infiltration.

(11) Feasible Ducting: The estimated percentage of exhaust air-to-supply air applications for which it would be economically feasible to run insulated ducting between the existing exhaust and supply duct locations. This value may also reflect unadaptable situations such as building side horizontal exhausters and cross-contamination restrictions.

(12) Extended Hours: The estimated percentage of exhausters which would be operated a minimum of 80 hours per week for 7 months of the year. This is a requirement for economic feasibility of the heat recovery unit.

(13) Exh. Air to Water: The estimated potential market for the exhaust air-to-water heat pump for the given building category.

(14) E. A. to Supply Air: The estimated potential market for the exhaust air-to-supply air heat pump or heat exchanger.

(15) Glycol Loop: The estimated potential market for the air-to-air heat pump with intermediate glycol loop, or for the run-around glycol loop.

Not all spreadsheets will contain all the lines, but only those required for complete calculations.

APPENDIX C REFERENCES

Statistics Canada Publications

1. No. SC81 92-932; 1981 Census; Table 7
2. No. 63-201; 1986; Table 1
3. Ibid.; Tables 2 and 4
4. No. 63-224; 1984; "Motion Picture Theatres"
5. No. 81-229; 1984; Tables 13 and 15
6. No. 63-210; 1984; Tables 1, 2, 3 and 8
7. No. 63-204; 1984; Tables 6 and 7
8. Ibid.; Tables 9 and 10
9. No. 85-211; 1984; Tables 7 and 24
10. No. 31-203; 1984; Tables 4, 28, 32, 48 and 52
11. No. 63-205; 1975; "Power Laundries, Dry Cleaning and Dyeing Plants"

BETT Publications

12. No. 82.02; "Building Category Profile: Warehouse and Light Industrial Buildings", Mar. 1982; Exhibits 3, 6 and 13.
13. Ibid.; Exhibits 8, 9, 10, 11 and 12.
14. No. 83.07; "Building Population Demographics and Energy Consumption for Health Care Facilities in Canada", Mar. 1983; Table C.2.
15. Ibid.; Table B.2.
16. No. 830.01; "Retail Stores: Energy Consumption and Potential Savings", Aug. 1983; Table 5.1.

Other Publications

17. Canadian Electrical Association Project 211 U SOA, "Hot Water Heat Pumps, Development Study", Jan. 1983; Figure 8.
18. Ontario Research Foundation, "Identification of the Energy Conservation Potential in the Medium to High-Rise Residential and Office Building Sector", Oct. 1982; Table 1, Series III A and III B.
19. Ibid.; Tables 2A and 2B.

20. Health and Welfare Canada, "National Study of Supply of Sports and Recreation Facilities - Final Technical Report on Phase II Socio-Cultural Facilities", Dec. 1974; Tables II and III.
21. Ibid. "- Final Technical Report on Phase III - Inventory of Sport Facilities", Dec. 1975; Tables XX and XXI.
22. Energy, Mi es and Resources, "Energy Statistics Handbook", Tables 1.1 and 1.15.
23. ASHRAE Publications, "ASHRAE Handbook; 1977 Fundamentals", Chapter 21, Table 6.

APARTMENT BUILDINGS

SIZE	<5 Storeys	>5 Storeys	TOTALS
No. of Units	1645636	920256	2565892
000's Sq. Ft.	1624473	908423	2532896
No. of Locations	63294	15598	78892
CFM Exhausted/Unit	50	50	
CFM Exhausted	8.228E7	4.601E7	
No. of Exhausters	41141	23006	64147
Water Demand	50%	100%	
Central Exhaust	10%	75%	
Feasible Ducting	50%	75%	
Extended Hours	70%	90%	
Exh. Air to Water	1440	15529	16969
E.A. to Supply Air	1440	11647	13087
Glycol Loop	2880	15529	18409

PENITENTIARIES & CORRECTIONAL FACILITIES

SIZE (Inmate Capacity)	<25	25 TO 199	>=200	TOTALS
No. of Locations	11	210	19	240
Exhausters/Location	0.5	4	8	
No. of Exhausters	6	840	152	998
Water Demand	0%	80%	100%	
Central Exhaust	30%	75%	75%	
Feasible Ducting	50%	75%	75%	
Extended Hours	80%	95%	100%	
Exh. Air to Water	0	479	114	593
E.A. to Supply Air	1	449	86	535
Glycol Loop	1	599	114	714

SPECIAL CARE FACILITIES

SIZE (Beds)	<25	25 TO 49	50 TO 199	>=200	TOTALS
No. of Locations	3201	922	1292	192	5607
No. of Beds	32478	20968	105165	77305	235916
CFM Exhausted/Bed	200	200	200	200	
CFM Exhausted	6495600	4193600	2.103E7	1.546E7	
No. of Exhausters	3248	2097	10517	7731	23592
Water Demand	30%	60%	100%	100%	
Central Exhaust	30%	50%	95%	95%	
Feasible Ducting	10%	20%	50%	50%	
Extended Hours	75%	95%	100%	100%	
Exh. Air to Water	219	598	9991	7344	18151
E.A. to Supply Air	73	199	4995	3672	8940
Glycol Loop	731	996	9991	7344	19061

HOSPITALS

SIZE (Beds)	<25	25 TO 49	>=50	TOTALS
No. of Locations	321	190	719	1230
No. of Beds	4815	6270	166707	177792
CFM Exhausted/Bed	300	300	300	
CFM Exhausted	1444500	1881000	5.004E7	
No. of Exhausters	722	941	25020	26683
Water Demand	30%	60%	100%	
Central Exhaust	80%	95%	100%	
Feasible Ducting	10%	15%	20%	
Extended Hours	100%	100%	100%	
Exh. Air to Water	173	536	25020	25729
E.A. to Supply Air	58	134	5004	5196
Glycol Loop	578	893	25020	26491

HOTELS
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SIZE (ROOMS)	<26	26 TO 199	>=200	TOTALS
No. of Locations	2052	1735	161	3948
No. of Rooms	30780	134129	66800	231709
CFM Exhausted/Room	30	30	30	
CFM Exhausted	923400	4023870	2004000	
No. of Exhausters	462	2012	1002	3476
Water Demand	30%	75%	90%	
Central Exhaust	30%	50%	75%	
Feasible Ducting	50%	75%	75%	
Extended Hours	50%	75%	90%	
Exh. Air to Water	21	566	609	1195
E.A. to Supply Air	35	566	507	1108
Glycol Loop	69	754	676	1500

MOTELS
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SIZE (Rooms)	<25	>=25	TOTALS
Nc of Locations	2767	992	3759
No. of Rooms	39605	47582	87187
CFM Exhausted/Room	25	25	
CFM Exhausted	990125	1189550	
No. of Exhausters	495	595	1090
Water Demand	10%	70%	
Central Exhaust	5%	25%	
Feasible Ducting	50%	50%	
Extended Hours	20%	50%	
Exh. Air to Water	0	208	209
E.A. to Supply Air	2	74	77
Glycol Loop	5	297	302

RETAIL STORES

TYPE	CHAIN	INDEPENDENT	TOTALS
No. of Locations	28412	58480	86892
000's Sq.Ft.	60646	124828	185474
CFM Exh'd/1000 S.F.	150	150	
CFM Exhausted	9096900	1.872E7	
No. of Exhausters	4548	9362	13911
Water Demand	5%	5%	
Central Exhaust	40%	40%	
Feasible Ducting	50%	50%	
Extended Hours	10%	20%	
Exh. Air to Water	9	37	47
E.A. to Supply Air	91	374	465
Glycol Loop	182	749	931

RESTAURANTS
MAINTENANCE

TYPE	LIC	UNLIC	DRIVE-IN TAKE-OUT	BARS/ CLUBS	CAFETERIA	REFRESH. STANDS	TOTALS
No. of Locations	10782	12834	6190	2846	1554	2472	36668
Exhausters/Location	1	1	0.5	0.5	0.5	0	
No. of Exhausters	10782	12834	3090	1423	777	0	28906
Water Demand	100%	100%	80%	90%	80%	10%	
Central Exhaust	95%	90%	70%	90%	70%	0%	
Feasible Ducting	70%	70%	95%	95%	80%	95%	
Extended Hours	80%	80%	80%	80%	30%	70%	
Exh. Air to Water	8194	9240	1384	820	131	0	17769
E.A. to Supply Air	5736	6468	1244	974	131	0	14952
Glycol Loop	8194	9240	1730	1025	163	0	20353

MOTION PICTURE THEATRES

	ALL	TOTALS
No. of Locations	1554	1554
Exhausters/Location	1.5	
No. of Exhausters	2331	2331
Water Demand	1%	
Central Exhaust	99%	
Feasible Ducting	90%	
Extended Hours	10%	
Exh. Air to Water	0	0
E.A. to Supply Air	199	199
Glycol Loop	221	221

OFFICE BUILDINGS

SIZE (Sq. Ft.)	<20000	20000 TO 49999	>=50000	TOTALS
No. of Locations	103	589	1111	1803
000's Sq. Ft.	1234	19421	180386	201041
CFM Exh'd/1000 S.F.	200	200	200	
CFM Exhausted	246800	3884200	3.608E7	
No. of Exhausters	123	1942	18039	20104
Water Demand	0%	5%	10%	
Central Exhaust	20%	50%	90%	
Feasible Ducting	50%	75%	75%	
Extended Hours	20%	20%	20%	
Exh. Air to Water	0	10	325	334
E.A. to Supply Air	2	146	2435	2583
Glycol Loop	5	194	3247	3446

WAREHOUSE:

SIZE (Sq. Ft.)	20000	50000 TO 149999	>=150000	TOTAL
No. of Locations	[illegible]	265	312	52109
DOU x Sq.Ft.	[illegible]	1855[illegible]	781[illegible]	497[illegible]
CFM Exh'd/1000 s.F.	10	10	10	
CFM Exhausted	[illegible]	1855050	781420	
No. of Exhausters	2170	928	391	3488
Water Demand	0%	0%	10%	
Centra. Exhaust	20%	30%	50%	
Feasible Ducting	0%	10%	30%	
Extended Hours	10%	10%	20%	
Exh. Air to Water	0	0	4	4
E.A. to Supply Air	0	3	12	15
Glycol Loop	42	28	39	110

LIGHT INDUSTRIAL

SIZE (Sq. Ft.)	<10000	10000 TO 49999	>=50000	TOTALS
No. of Locations	6795	9413	2927	19135
000's Sq. Ft.	24814	203390	214911	443115
CFM Exh'd/1000 S.F.	200	200	200	
CFM Exhausted	4962800	4.068E7	4.298E7	
No. of Exhausters	2481	20339	21491	44312
Water Demand	20%	30%	40%	
Central Exhaust	10%	40%	60%	
Feasible Ducting	50%	50%	50%	
Extended Hours	30%	30%	30%	
Exh. Air to Water	15	732	1547	2294
E.A. to Supply Air	37	1220	1934	3192
Glycol Loop	74	2441	3868	6384

MANUFACTURING INDUSTRY

SIZE (Employees)	<50	50 TO 199	>=200	TOTALS
No. of Locations	30689	5192	1645	37526
Exhausters/Location	0.2	3	6	
No. of Exhausters	6138	15576	9870	31584
Water Demand	20%	30%	40%	
Central Exhaust	30%	80%	80%	
Feasible Ducting	50%.	50%	50%	
Extended Hours	30%	30%	30%	
Exh. Air to Water	110	1121	948	2179
E.A. to Supply Air	276	1869	1184	3330
Glycol Loop	552	3738	2369	6659

ELEMENTARY & SECONDARY SCHOOLS

SIZE (Enrolment)	<200	200 TO 999	>=1000	TOTALS
No. of Locations	6318	8632	569	15519
No. of Students	550150	3493950	816900	4861000
CFM Exh'd/Student	5	10	10	
CFM Exhausted	2750750	3.494E7	8169000	
No. of Exhausters	1375	17470	4085	22930
Water Demand	20%	60%	90%	
Central Exhaust	20%	75%	95%	
Feasible Ducting	50%	50%	50%	
Extended Hours	20%	40%	40%	
Exh. Air to Water	11	3145	1397	4552
E.A. to Supply Air	28	2620	776	3424
Glycol Loop	55	5241	1552	6848

UNIVERSITIES & COMMUNITY COLLEGES

SIZE (Enrolment)	<300	300 TO 2999	3000 TO 9999	>=10000	TOTALS
No. of Locations	75	106	61	18	260
Exhausters/Location	1	3	10	20	
No. of Exhausters	75	318	610	360	1363
Water Demand	30%	50%	60%	55%	
Central Exhaust	80%	90%	100%	85%	
Feasible Ducting	80%	80%	70%	55%	
Extended Hours	50%	70%	80%	75%	
Exh. Air to Water	9	100	293	126	528
E.A. to Supply Air	24	160	342	126	652
Glycol Loop	30	200	488	230	948

[illegible]

	ALL	[illegible]
No. of Locations	3287	3287
Exhausters/Location	1.5	
No. of Exhausters	5094	5094
Water Demand	100%	
Central Exhaust	90%	
Feasible Ducting	30%	
Extended Hours	75%	
Exh. Air to Water	3439	3439
E.A. to Supply Air	2752	2752
Glycol Loop	3439	3439

RECREATIONAL FACILITIES 1

TYPE	LIBRARY	MUSEUM	GYMS.	SQUASH/ HANDBALL	TOTALS
No. of Locations	289	1609	8834	236	10968
Exhausters/Location	1	3	6	3	
No. of Exhausters	289	4827	53004	708	58828
Water Demand	0%	0%	90%	90%	
Central Exhaust	90%	90%	95%	75%	
Feasible Ducting	90%	90%	90%	90%	
Extended Hours	60%	40%	80%	80%	
Exh. Air to Water	0	0	36255	382	36637
E.A. to Supply Air	140	1564	36255	382	38341
Glycol Loop	156	1738	40283	425	42602

RECREATIONAL FACILITIES 2

TYPE	INDOOR HOCKEY RINKS	BOWLING LANES	CURLING RINKS	INDOOR SWIMMING POOLS	TOTALS
No. of Locations	1570	1273	884	1120	4847
Exhausters/Location	3	2	2	2	
No. of Exhausters	4710	2546	1768	2240	11264
Water Demand	70%	0%	30%	100%	
Central Exhaust	95%	90%	95%	100%	
Feasible Ducting	50%	50%	50%	50%	
Extended Hours	60%	70%	60%	80%	
Exh. Air to Water	1879	0	302	1792	3974
E.A. to Supply Air	1342	802	504	896	3544
Glycol Loop	2685	1604	1008	1792	7088